International Journal of Engineering Research (ISSN : 2319-6890)
Volume No.2, Issue No. 7, pp : 467-471 01 Nov 2013

Mathematical Modeling of an Automobile Damper

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Abstract - ***In an automotive industry, to reduce product development time and increase quality of product, it is essential to reduce the number of physical prototypes and rely more on precise & reliable design for the final design of vehicles. This paper presents a mathematical model for the damping force of the hydraulic shock absorber which is implemented to analyse the shock absorbers mounting brackets attached to the vehicle structure. Physical testing results indicate that the considered shock absorber's mathematical model is reliable and can be used to calculate the durability target life of mounting brackets. Thus this presented methodology can be utilized as an effective way to reduce time and cost in design and development of automotive components.***

Key words - Shock absorber, Mathematical Modeling, Damping Force, Damper Test Rig, Durability

I. Introduction

In this presented work, the methodology for the damping force analysis of an automobile shock absorber using a mathematical model is proposed. In a vehicle, shock absorbers reduce the effect of sudden displacements occurring due to passing over rough ground. This leads to improved ride quality and increase in comfort. While shock absorbers serve the purpose of limiting excessive suspension movement, their intended sole purpose is to dampen spring oscillations.

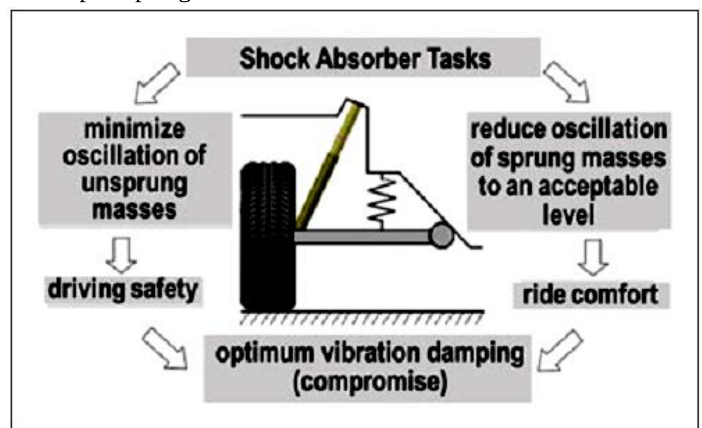


Fig. 1.1: The Shock Absorber: Purposes and Tasks [1]

Much effort has been made by numerous researchers to develop mathematical models that allow the hysteretic behaviour of dampers. Identification approaches can be divided into two categories: parametric and non-parametric. A mathematical model for the Magnetorheological (MR) damper using parametric modellingapproach was developed [2]. When compared with experimental data, it was shown that the resulting model accurately predicts the response of the MR damper over a wide range of operating conditions. The influence of damper properties on vehicle behaviour was studied [3]. The non-linear hysteretic physical shock absorber model and the processes utilized to identify the constituent parameters were discussed.An experimentally validated physical model for a high-performance gas-charged Mono-Tube racing damper was developed [4]. The model includes bleed orifice, piston leakage, and shim stack flows. The model is validated with experimental tests on an Ohlins WCJ 22/6 damper and shown to be accurate. The influence of a shock absorber models on vehicle dynamic simulation was studied [5]. The real behaviour of a European medium-range car shock absorber has been obtained by means of a test rig. From the damper's real behaviour, three mathematical models were generated, increasing the complexity. An existing full vehicle simulation application (CarSim) was used for this particular study.

In summary, various researchers as mentioned above have proposed the suitable mathematical models for damper as per the application field and with the help of testing of damper under physical and/or virtual simulation. Next section describes the methodology for damping force analysis of shock absorber. In the last section, the analytical results are compared with the physical testing results to derive the conclusions.

II. Methodology

Shock absorber is an important part of automotive which has an effect on ride characteristics such as ride comfort and driving safety. There are several kinds of automotive shock dampers such as position-sensitive damping, acceleration-sensitive damping, and continuous damping control. Displacement-sensitive shock absorber (DSSA), which is also called stroke-dependent shock absorber, and has a similar structure compared with conventional passive shock absorber. Damping characteristics of automotive can be analyzed by considering the performance of displacement-sensitive shock absorber (DSSA) for the ride comfort.

Functions of Shock Absorbers in Vehicles

Vibration dampers are arranged parallel to the vehicle suspension and have the following tasks: to dampen vibrations

of the vehicle's body caused by uneven roads or driving conditions and to quickly reduce and eliminate road-induced wheel and axle vibration in order to provide constant contact between the tire and the roadway.This helps ensure good tracking and braking performance.

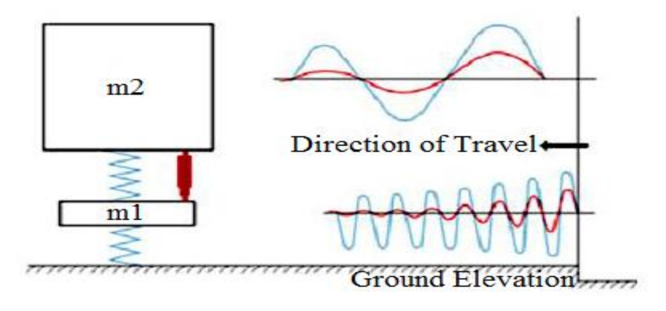


Fig. 3.1: Coupled Chassis and Body Mass Vibrations

When the vehicle passes over a bump, the suspension and vibration dampers are compressed. The resulting shock of the vehicle is absorbed by the suspension. The suspension prevents the sprung mass (body and payload) from making contact between the unsprung mass (suspension and wheels). The springs, however, tend to relax again, thereby releasing the energy stored within them. In order to quickly reduce and eliminate this springing oscillation between axle and body, the chassis is equipped with vibration dampers. Sprung and unsprung masses vibrate in different frequency ranges. The graphs in Fig. 1.4 clearly illustrate how the vibrations caused by a roadway input (undamped vibrations: lighter curves) are reduced by the damper (darker curves) [1].

Construction:

Consider a hydraulic, single tube, telescopic damper. Fig. 3.2 depicts a typical, NASCAR, single tube damper. The mono-tube damper is the preferred construction in racing applications.

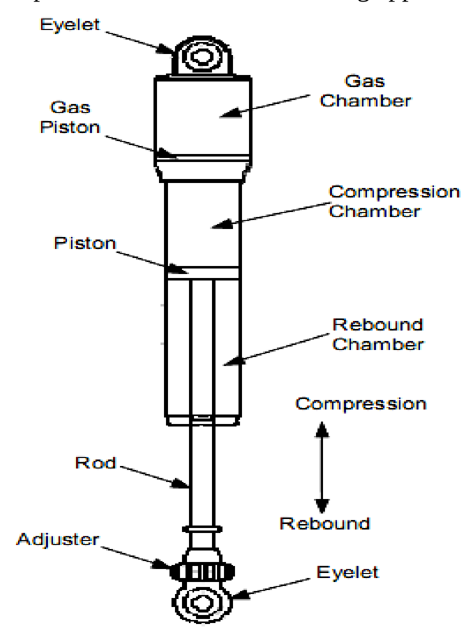


Fig. 3.2: Major Components of a Mono-Tube Damper [7]

The damper consists of three chambers: gas, rebound and compression. The gas chamber is at the top of the tube; it is separated from the compression chamber by a floating piston. This piston separates the gas in the gas chamber, typically nitrogen, from the oil in the compression chamber. The compression chamber sits between the floating piston of the gas chamber and the piston. The rebound chamber is opposite the compression chamber on the other side of the piston and at the bottom end of the tube. Both the compression chamber and rebound chamber are completely filled with high-quality oil. The rod passes through a special seal designed to keep the oil in, dirt out and to minimize friction between the rod and seal. The damper is attached to the vehicle through two eyelets[7].

The damper operates in two modes, compression (positive velocity) and rebound (negative velocity). During the compression stroke the rod is pushed into the tube and fluid flows through the piston from the compression chamber to the rebound chamber.

Fluid Flow passages and Working:

The main mechanism for providing damping is by shearing the hydraulic fluid as it flows through restrictions. This dissipates energy by generating heat in the fluid that is then dissipated to the shock tube and then to the atmosphere. The other mechanism for damping is friction between the various moving parts of the damper.

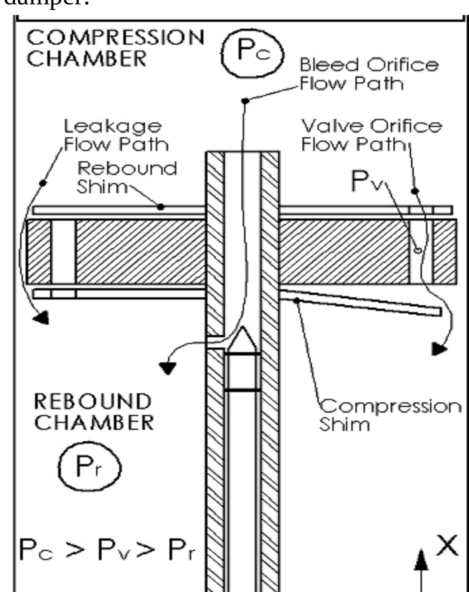


Fig. 3.3: Compression Stroke Flow Diagram [7]

Total fluid flow is split among three possible paths. The first is through the bleed orifice located in the end of the rod (See label 3 in Fig. 3.4 and 3.6). Fluid can flow through this orifice at all piston speeds from the compression to rebound chamber and vice versa. The bleed orifice dominates the low speed characteristics of the damper. The area of the orifice can be adjusted by screwing a needle valve in and out.

The second fluid flow passage is through the rebound or compression valves on the piston. Label (1) in Fig. 3.4 and label (2) in Fig. 3.6 depict fluid flow through the rebound valve

and compression valve respectively. These valves are essentially check valves that allow flow in only one direction. Each valve consists of an orifice in the piston and a shim stack. The shim stack is a series of thin circular steel discs stacked according to diameter.Fig. 3.4 shows the rebound shim stack and rebound piston orifice. The combination of the piston orifice and the annular flow path created by deflecting the shim stack puts two flow resistances in series.

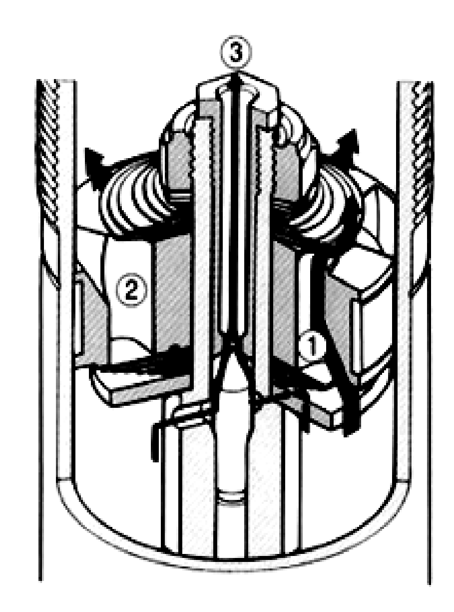


Fig. 3.4: Cutaway Showing Fluid Flow Paths through the Rebound Piston Orifice and Shim Stack (1) and Through the Bleed Orifice (3), During the Rebound Stroke [7]

The shim stack blocks the piston orifice unless a pressure differential exists. The stacks are typically preloaded by dishing the piston slightly. This preload prevents the stack from opening until the pressure differential reaches a desired level.

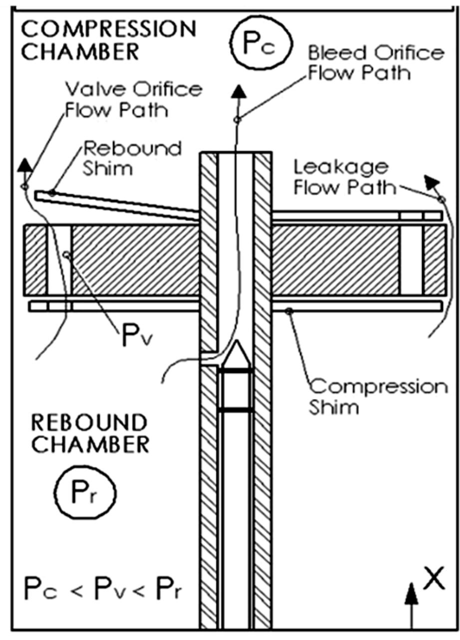


Fig. 3.5: Rebound Stroke Flow Diagram [7]

The compression valve is located on the rebound side of the piston. Fig. 3.6 shows the flow paths during the compression stroke. The final flow path is leakage between the piston ring and tube wall.

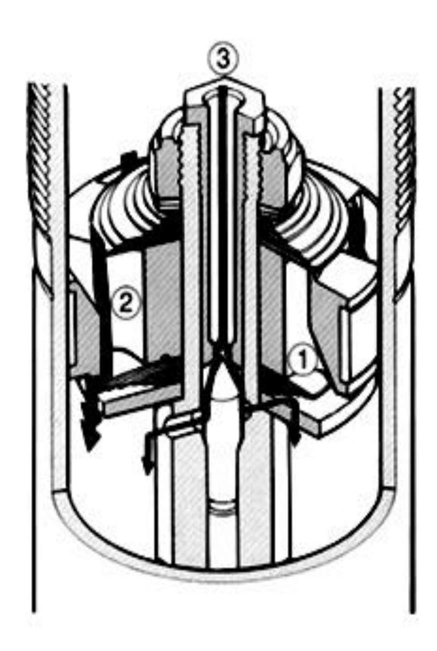


Fig. 3.6: Cutaway Showing Fluid Flow Paths through the Compression Piston Orifice Shim Stack (2) and Through the Bleed Orifice (3), During Compression Stroke [7]

A floating piston separates the compression chamber from the gas chamber. The gas chamber contains pressurized gas, usually air or nitrogen. Dried nitrogen is preferred because it is more stable with temperature changes due to the lack of water vapor.

This pressurized gas chamber keeps the oil in the damper pressurized to prevent cavitations. The gas chamber also accounts for the volume of the rod entering and exiting the tube during piston motion. As the rod enters the tube during compression, the gas will compress and the floating piston will move up to decrease the gas volume by the amount of piston rod volume that has entered the damper body. When the rod is drawn out of the tube, the gas expands and the floating piston moves down. The pressure in the gas chamber also gives the damper a small gas spring effect.

Damping Force Analysis:

The compression stroke in the damper is shown in Fig. 3.7 considering general fluid flow. It shows three paths of fluid flow. The compression valve is open allowing flow through the piston. The bleed orifice is also allowing fluid flow and there is a small amount of leakage past the piston and cylinder wall. The gas piston movement (z) is proportional to the amount of rod insertion (x). The rebound stroke is the reverse of this compression stroke, with the rebound valve on the compression side of the piston being open and the compression valve closed. Throughout the following mathematical model development the damper is having compression stroke.

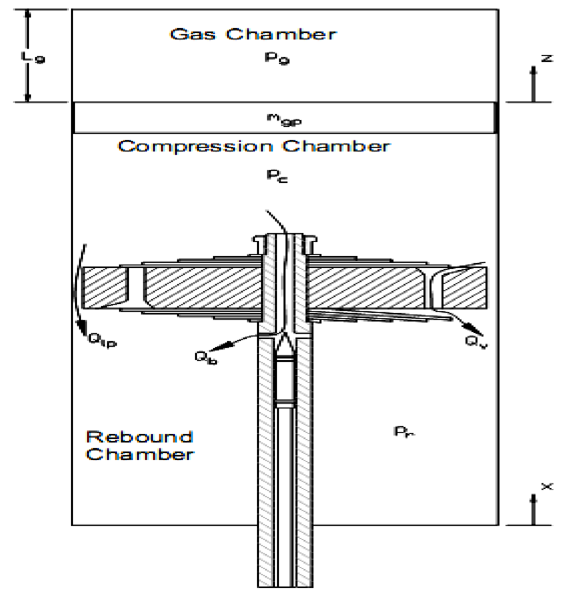


Fig. 3.7: Compression Stroke Diagram Showing the Three Flow Paths [7]

To determine the force the damper produces for a given speed, a free-body diagram is constructed for the piston and rod assembly as shown in Fig 3.8. This is the basis for finding the damping force exerted by the damper. The forces acting on the piston-rod assembly during a cycle are: 1) the pressure force differential across the piston (p_rA_r- p_cA_c) and 2) the friction force F_f between the piston ring and tube and between the rod and the seal.

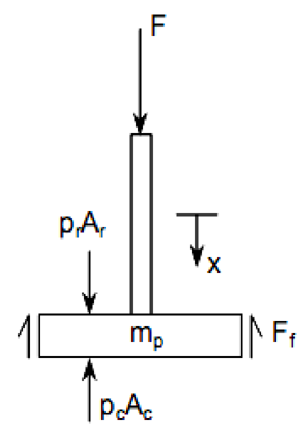


Fig. 3.8: Free-Body Diagram of Piston-rod Assembly [7]

Summing the forces on the piston in the direction of displacement x gives:

$$F + p_r A_r - p_c A_c - F_f = m_p \ddot{x} \quad \text{...(1)}$$

This equation can be used to solve for the damper force F, as a function of damper motion (position, velocity, and acceleration).

3.2.1 General Damper Configuration

A general damper configuration is shown in the Fig. 3.9, the packaging may be different from this, but the operating principle is the same. The Fig. 3.9 shows a remote reservoir with a free piston separating gas and liquid, but the gas may be accommodated in an alternative way, e.g. around the main cylinder in the case of the double-tube damper. The piston has two valves, PE (piston extension) and PC (piston compression). Entry or exit of the rod causes fluid displacement through one of the two foot valves, FC (foot compression) and FE (foot extension)respectively.

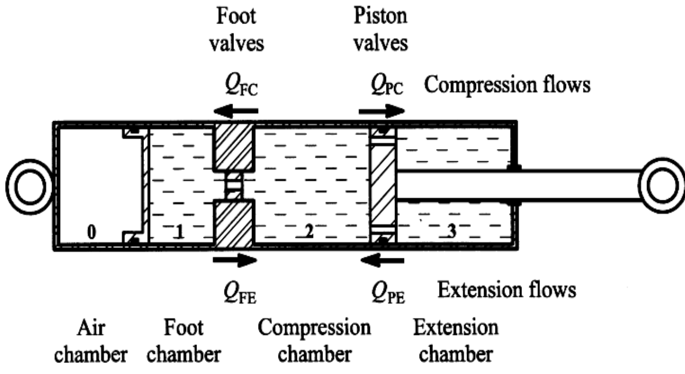


Fig. 3.9: General Damper Configuration [7]

Consider the damper to be displaced by a small distance X_{DC} inwards. The volume of the compression chamber (Chamber 2) is reduced, resulting in an increased pressure during the motion to force liquid out, some going to the expansion chamber (Chamber 3) and some to the foot chamber (Chamber 1).

Consider,

A_R = area of the rod moving through extension chamber region,

A_P = area of the piston moving through hydraulic chamber region

A volume V_{FC} of liquid is displaced by the rod and is moved through the foot valve:

$V_{FC} = A_R X_{DC}$

where X_{DC} = damper rod displacement

The volume moved through the piston compression valve is

$V_{PC} = (A_P – A_R)X_{DC}= A_{PA}X_{DC}$

For a damper compression velocity V_{DC}, the volumetric oil flow rates (assuming correct operation and incompressible liquid) in the damper in different strokes are as below

- In compression stroke
  $Q_{FC} = A_RV_{DC},\quad Q_{PC} = A_{PA}V_{DC}$
- Similarly , in extension stroke
- $Q_{FE} = A_RV_{DE},\quad Q_{PE} = A_{PA}V_{DE}$
  The chamber pressures are calculated as below
- In compression stroke,

foot valve pressure drop $P_{FC} = k_{FC}Q_{FC}$

piston valve pressure drop $P_{PC} = k_{PC}Q_{PC}$

- In extension stroke,
  foot valve pressure drop $P_{FE} = k_{FE}Q_{FE}$
  piston valve pressure drop $P_{PE} = k_{PE}Q_{PE.}$
  The different valve resistances are calculated as
- In compression stroke,
  piston valve resistance $k_{FC}= \frac{C_{DC}}{A_{PA}A_P}$
  foot valve resistance $k_{FC} = \frac{C_{DC}}{A_R A_P}$
- In rebound stroke ,

piston valve resistance $k_{PE} = \frac{C_{DE}}{A_{PA}}$

foot valve resistance $k_{FE} = 0$.

So,

- In compression stroke,

Compression chamber pressure $P_c = P_{FC}$

Rebound chamber pressure $P_r = P_c - P_{PC}$

- In rebound stroke,

Compression chamber pressure $P_c = P_{FE}$

Rebound chamber pressure $P_r = P_c - P_{PE}$

IV. Results and Discussion

Sample Calculation for the Damper under consideration

- Rod diameter = 0.013 m, Piston diameter = 0.041 m, $A_c = 1.32 \times 10^{-3}\ m^2$ $A_r = 6.157 \times 10^{-4}\ m^2$

In compression stroke, for velocity = 0.52 m/s

- $k_{FC} = 1.14178 \times 10^{10}\ Ns/m^5$ $k_{PC} = 2.460 \times 10^{9}\ Ns/m^5$
- $P_{FC} = 787818.668\ N/m^2$ $P_{PC} = 787603.44\ N/m^2$
- $P_r = 215.228\ N/m^2$ $P_c = 787818.668\ N/m^2$
- $\ddot{x} = 0.08639\ m/s^2$

Damping force, F = 1074.2636 N

Testing Results for damper:

The shock absorber is physically tested to find out the damping forces for different damper rod velocities using damper test rig. Two important conclusions are drawn from the damper testing:

- Damping force *F* is a non-linear function of damper rod speed *v*
- Relation is unsymmetrical across rebound and compression Region.

Table 6.1 Testing Results for damper

Damper Rod Velocity m/s	Compression Force N	Rebound Force N
0.06	173.637	585.76
0.10	439.49	1523.49
0.21	699.45	3407.01
0.31	802.46	4063.3
0.42	894.67	4450.79
0.54	977.08	4814.75
0.63	988.85	5037.44
1.01	1239.01	6020.41
1.50	1514.67	7162.28
2.00	1874.69	8193.31
2.51	2201.36	9213.55
3.01	2353.42	10452.56

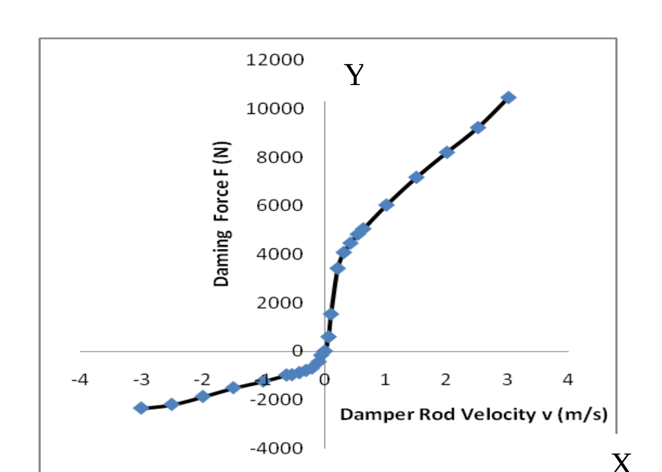


Fig. 6.2: Damping Force (N) vs Damper Rod Velocity (m/s)

These testing results can be used to calculate durability load cases in further development.

V. Conclusion

- The results obtained by experimental method using damper test rig are close to results obtained by analytical model of damping force with 10 % of error. The errors in the results are due to difference in the loading conditions, the mounting assembly of the damper and the frictional force between moving parts.
- Therefore, these observations confirm that mathematical model is reliable and precise.

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